



[001] AUTOMATICALLY SHIFTABLE MOTOR VEHICLE TRANSMISSION

[002]

[003] The present invention concerns an automatically shiftable motor vehicle transmission of planetary design, according to the preamble of Claim 1.

[004]

[005] According to the existing art, automatic transmissions, in particular for motor vehicles, usually encompass planetary gear sets that are shifted by means of frictional or shifting elements, for example clutches and brakes, and usually are connected to a initial movement element, for example a hydrodynamic torque converter or a fluid coupling, that is based on a slip effect and is selectably equipped with a lockup clutch.

[006] A transmission of this kind is disclosed in EP 0 434 525 B1. It substantially encompasses an input drive shaft and an output drive shaft that are arranged parallel to one another; a double planetary gear set arranged concentrically with the output drive shaft; and five shifting elements in the form of three clutches and two brakes, the selectable locking of which, in respective pairs, determines the various gear ratios between the input drive shaft and output drive shaft. The transmission has a reduction gear set and two power paths, so that by selective paired engagement of the five shifting elements, six forward speeds are obtained.

[007] For the first power path, two clutches are needed in order to transfer torque from the reduction gear set to two elements of the double planetary gear set. These are arranged in the power flow direction substantially after the reduction gear set in the direction of the double planetary gear set. For the second power path, a further clutch is provided that releasably connects it to a further element of the double planetary gear set. The clutches are arranged in such a way that the inner disc carrier constitutes the output drive.

[008] A further planetary transmission of this kind is known, for example, from U.S. Pat. No. 4,070,927, the number of available forward gears being in each case one greater than the number of frictional or shifting elements. Each gear change between the forward gears is achieved by respectively switching one of the frictional or shifting elements in or out.

- [009] Also known, from DE 199 12 480 A1 of the present inventor, is an automatically shiftable motor vehicle transmission having three single-carrier planetary gear sets as well as three brakes and two clutches for selecting six forward gears and one reverse gear, in which the input drive shaft is connected directly to the sun gear of the second planetary gear set and is connectable via the first clutch to the sun gear of the first planetary gear set and/or via the second clutch to the carrier of the first planetary gear set. Additionally or alternatively, the sun gear of the first planetary gear set is connectable via the first brake to the housing of the transmission, and/or the carrier of the first planetary gear set is connectable via the second brake to the housing, and/or the sun gear of the third planetary gear set is connectable via the third brake to the housing.
- [010] Also described in EP 1 265 006 A2 is a six-speed automatic transmission of planetary design which also encompasses two brakes and two clutches, in which the individual gear ratios can be implemented by a combination of two shifting elements in each case.
- [011] Automatically shiftable vehicle transmissions of planetary design in general have already been described many times in the existing art, and are continuously being developed and improved. For example, these transmissions are intended to have a sufficient number of forward speeds as well as a reverse gear, as well as a conversion ratio that is highly suitable for motor vehicles, with a wide overall ratio spread and favorably spaced ratios. They should also make possible a high initial movement conversion ratio in the forward direction and contain a direct ratio, and be suitable for use in both passenger cars and commercial vehicles. These transmissions should also require little design complexity, in particular a small number of shifting elements; and should prevent double shifts upon sequential shifting, so that when shifts are made in defined speed groups, only one shifting element is changed in each case.
- [012] Transmissions of the kind described above generally have six forward speeds; a further, seventh speed would advantageously result in more convenience for the driver and in optimized fuel consumption. The seventh gear also optimally guarantees a sporty driving style.

[013] It is the object of the present invention to propose a multiple-ratio transmission of the kind described above in which at least seven forward speeds can be implemented. It is intended, in the multiple-ratio transmission according to the present invention, that low torques act on the shifting elements and planetary gear sets; and that the rotation speeds of the shafts, shifting elements and planetary gear sets be minimized. The transmission according to the present invention is furthermore intended to be light in weight with compact dimensions. Any desired embodiment and arrangement of an initial movement element, and of the input drive and output drives, should also be possible.

[014] This object is achieved, according to the present invention, by the features of Claim 1. Further advantages and advantageous embodiments are evident from the dependent claims.

[015]

[016] In accordance therewith, an automatically shiftable multiple-ratio transmission of planetary design is proposed, encompassing an input drive shaft and an output drive shaft that are arranged in a housing; three single-carrier planetary gear sets; at least six rotatable shafts; and at least six shifting elements, preferably encompassing three brakes and three clutches, selective engagement of which (in pairs) brings about various conversion ratios between the input drive shaft and output drive shaft, so that at least seven forward speeds and at least one reverse gear can be implemented, in which the input drive shaft is continuously connected to the sun gear of the second planetary gear set, is connectable via a clutch to the carrier of the first planetary gear set, and is connectable via a further clutch to a fifth shaft which on the one hand is continuously connected to the sun gear of the first planetary gear set and on the other hand is couplable via a brake to the housing. In addition, the output drive shaft is continuously connected to the carrier of the third planetary gear set and to the ring gear of the first planetary gear set, a further third shaft being continuously connected to the sun gear of the third planetary gear set and couplable by way of a brake to the housing.

- [017] Furthermore, according to the present invention, a further, fourth shaft is continuously connected to the ring gear of the second planetary gear set and to the carrier of the first planetary gear set, and is couplable via a brake to the housing; a further, sixth shaft is continuously connected to the ring gear of the third planetary gear set and to the carrier of the second planetary gear set. According to the present invention, a further clutch is provided which releasably interconnects the fourth shaft to the output drive shaft or to the fifth shaft. The single-carrier planetary gear sets are preferably embodied as minus planetary gear sets.
- [018] The configuration according to the present invention of the multiple-ratio transmission results in suitable conversion ratios as well as a considerable increase, as compared with the existing art, in the overall ratio spread of the multiple-ratio transmission; this brings about an improved adaptation, over the entire operating range of the motor vehicle, of the gear ratios to the respective rotational speeds of the drive engine to which the multiple-ratio transmission is connected, with the consequence of improved drivability of the motor vehicle and a significant decrease in fuel consumption. Comparatively favorably spaced ratios and a high initial movement conversion ratio in the forward direction can moreover be achieved.
- [019] The multiple-ratio transmission according to the present invention is suitable for any motor vehicle, in particular for passenger cars and for commercial vehicles, e.g., trucks, buses, construction vehicles, rail vehicles, tracked vehicles, and the like.
- [020] With the multiple-ratio transmission according to the present invention the design complexity is moreover considerably reduced as a result of a small number of shifting elements, preferably three clutches and three brakes. With the multiple-ratio transmission according to the present invention it is advantageously possible to perform an initial movement using a hydrodynamic converter, an external initial movement clutch, or also other suitable external initial movement elements. It is also conceivable to enable an initial movement operation using an initial movement element integrated into the transmission.

[021] According to the present invention, one gear, preferably the sixth gear, can be designed as a direct ratio; double shifts and group shifts upon sequential shifting are moreover avoided.

[022] The input drive shaft and output drive shaft can be arranged both coaxially with one another on opposite sides, and also both on the same side, of the transmission housing. It is also possible to arrange the output drive between the planetary gear sets and the clutches.

[023]

[024] Further goals, features, advantages, and potential applications of the invention are evident from the description below of the exemplary embodiments, which are depicted in more detail in the drawings. All features described and/or visually depicted, of themselves or in any appropriate combination, constitute the subject matter of the invention, irrespective of their grouping in the claims and their internal references. In the drawings:

[025] Fig. 1 is schematically depicts a first embodiment of a transmission according to the present invention;

[026] Fig. 2 shows, by way of example, a shifting diagram for the transmission depicted in Fig. 1;

[027] Fig. 3 is schematically depicts a second embodiment of a transmission according to the present invention;

[028] Fig. 4 shows, by way of example, a shifting diagram for the transmission depicted in Fig. 2;

[029] Fig. 5 is schematically depicts a variant component arrangement for the transmission depicted in Fig. 1; and

[030] Fig. 5 is schematically depicts a variant component arrangement for the transmission depicted in Fig. 3.

[031]

[032] Identical components in different Figures, or components having identical functions, are labeled in the Figures with the same reference characters.

[033] A transmission according to the present invention substantially encompasses, according to FIG. 1, three single-carrier planetary gear sets P1, P2, P3 embodied as minus planetary gear sets; six rotatable shafts 1, 2, 3, 4, 5, 6; three brakes 03, 04, 05; and three clutches 14, 15, 45. All these components are arranged inside a housing G. Planetary gear sets P1, P2, P3 each comprise a sun gear, a carrier with planetary gears, and a ring gear. The three planetary gear sets P1, P2, P3 are arranged coaxially with one another, the middle planetary gear set (viewed three-dimensionally) being labeled P2. In the context of the exemplary embodiment shown in FIG. 1, the planetary gear sets are connected to one another as set forth below.

[034] Input drive shaft 1 is continuously connected to the sun gear of second planetary gear set P2, and is connectable via a clutch 14 to the carrier of first planetary gear set P1; input drive shaft 1 is furthermore connectable via a further clutch 15 to shaft 5, which on the one hand is continuously connected to the sun gear of first planetary gear set P1 and on the other hand is couplable via a brake 05 to housing G. Output drive shaft 2 is continuously connected to the carrier of third planetary gear set P3 and to the ring gear of first planetary gear set P1, and a further shaft 3 is continuously connected to the sun gear of third planetary gear set P3 and couplable via a brake 03 to housing G. A further shaft 4 is continuously connected to the ring gear of second planetary gear set P2 and to the carrier of first planetary gear set P1, and is couplable via a brake 04 to housing G. Shaft 6 is furthermore continuously connected to the ring gear of third planetary gear set P3 and to the carrier of second planetary gear set P2. According to the invention, a clutch 45 is provided which releasably connects shaft 4 to shaft 5. Seven forward speeds are implemented by the provision of this internal clutch.

[035] Shaft 4 is releasably connected, in a region between brake 04 and the carrier of first planetary gear set P1, to shaft 5 via clutch 45.

[036] Fig. 2 depicts a shifting diagram of the multiple-ratio transmission according to the present invention shown in FIG. 1. The shifting diagram illustrates, by way of example, the respective conversion ratios i of the individual gear ratios, and the

ratio intervals to be determined therefrom. It is also evident from the shifting diagram that double shifts or group shifts can be avoided in the context of sequential shifting because each two adjacent gear ratios use a common shifting element. As is apparent from the shifting diagram, it is also possible to skip gears without group shifting. The sixth gear is preferably embodied as a direct ratio.

[037] For the first five speeds, brake 03 is continuously activated; in addition, brake 04 is closed for first gear, brake 05 for second gear, clutch 45 for third gear, clutch 15 for fourth gear, and clutch 14 for fifth gear. Clutches 14 and 15 must be closed for sixth gear, and clutch 14 and brake 5 for seventh gear. For the reverse gear, brake 04 and clutch 15 are closed.

[038] The spatial arrangement of the shifting elements in the transmission according to the present invention can be any one desired; it is limited only by dimensions and external configuration. In conjunction with a non-coaxial - in particular, axially parallel - arrangement of input drive AN and output drive AB of the transmission, clutches 14, 15, 45 are preferably arranged, when viewed radially, above planetary gear sets P1, P2 and P3. In the embodiment shown by way of example in Fig. 1, clutch 45 is arranged, when viewed radially, slightly above the first (input-drive-side) planetary gear set P1, and closer thereto than clutches 14 and 15. The inner disc carrier of clutch 45 is preferably arranged on the output drive side; a servo device (not depicted, for simplification) of clutch 45 can be arranged between the disc packet and the outer disc carrier. The outer disc carriers of clutches 14 and 15 are preferably also arranged on the input drive side.

[039] The transmission configuration as depicted in Fig. 3 corresponds substantially to the embodiment shown in Fig. 1, with the difference that shaft 4 is releasably connectable to drive shaft 2, via clutch 24, in a region between brake 04 and the carrier of first planetary gear set P1. In this embodiment, the inner disc carrier of clutch 24 is preferably arranged on the output drive side.

[040] It is also clearly evident from Fig. 3 that by simultaneously closing brake 04 and clutch 24 it is easy to constitute a so-called "hill holder" for the transmission, with which output drive shaft 2 of the transmission can be locked toward housing G.

- [041] Fig. 4 depicts a shifting diagram of the multiple-ratio transmission according to the present invention shown in Fig. 3. The only difference as compared with the shifting diagram shown in Fig. 2 is that instead of clutch 45, clutch 24 is provided and actuated (third gear).
- [042] For the first five speeds, brake 03 is continuously activated; in addition, brake 04 is closed for first gear, brake 05 for second gear, clutch 24 for third gear, clutch 15 for fourth gear, and clutch 14 for fifth gear. Clutches 14 and 15 must be closed for sixth gear, and clutch 14 and brake 5 for seventh gear. For reverse gear, brake 04 and clutch 15 are closed.
- [043] While the two embodiments according to the present invention described above each show an axially parallel arrangement of input drive AN and output drive AB of the transmission, a description will now be given, for each of the two embodiments, of a variant component arrangement in which input drive AN and output drive AB are arranged coaxially with one another.
- [044] Fig. 5 schematically depicts an example of a variant component arrangement for the transmission depicted in Fig. 1. The kinematic coupling of the three planetary gear sets P1, P2, P2 and the six shifting elements 03, 04, 05, 14, 15, 24 is thus carried over without change from Fig. 1. As already mentioned, in contrast to Fig. 1 input drive AN and output drive AB of the transmission now run coaxially with one another. Input drive AN of the transmission is therefore arranged on the side of first planetary gear set P1 facing away from the second (middle) planetary gear set P2, and output drive AB of the transmission is located on the side of the transmission opposite input drive AN, i.e., on the side of third planetary gear set P3 facing away from the second (middle) planetary gear set P2. Clutch 14, with which shaft 4 is connectable to input drive shaft 1, is now arranged axially between the two planetary gear sets P1 and P2. The other two clutches 15 and 45, and also the two brakes 04 and 05, are arranged on the input drive side of the transmission, i.e., on the side of first planetary gear set P1 facing away from middle planetary gear set P2. Brake 05 is arranged adjacent to the input-drive-side outer wall of housing G. The two clutches 15 and 45 are arranged adjacent to one another, clutch 45 being arranged closer to planetary gear set P1 than is clutch 15.

In the example depicted in Fig. 5, clutch 45 and brake 04 are arranged in three-dimensional terms adjacent to first planetary gear set P1.

[045] In another embodiment of the component arrangement, provision can also be made, for example, for brake 04 to be arranged in three-dimensional terms in a region radially above the planetary gear sets, in particular radially above first planetary gear set P1. Clutch 45 can also be arranged below brake 04 (viewed in the radial direction). In yet another embodiment of the component arrangement, clutches 15 and 45 can also be grouped together as a pre-assemblable subassembly having one common disc carrier. Clutch 15 can also be arranged in three-dimensional terms radially above brake 05, or brake 05 (e.g. in the form of a band brake) can be arranged in three-dimensional terms radially above clutch 15.

[046] One skilled in the art can adapt the three-dimensional arrangement of the respective shifting elements to the installation space available for the transmission in the motor vehicle.

[047] Lastly, Fig. 6 schematically depicts an example of a variant component arrangement for the transmission depicted in Fig. 3. The kinematic coupling of the three planetary gear sets P1, P2, P2 and the six shifting elements 03, 04, 05, 14, 15, 24 is thus carried over without change from Fig. 3. As already mentioned, in contrast to Fig. 3 input drive AN and output drive AB of the transmission run coaxially with one another. In addition, all the shifting elements (04, 05, 14, 15, 24) with the exception of brake 03 are now arranged on the input drive side of the transmission, i.e., on the side of first planetary gear set P1 that is located opposite second planetary gear set P2. Clutch 24, with which output drive shaft 2 is additionally connectable to shaft 4, is adjacent to first planetary gear set P1. The two clutches 14 and 15, in particular their disc packets, are arranged next to one another, clutch 14 being adjacent to the input-drive-side outer wall of housing G. The two clutches 14, 15 can also be grouped together as a pre-assemblable subassembly, in particular having one common disc carrier. In another embodiment, provision can also be made, for example, for the two clutches 14, 15 to be arranged nested one inside another, clutch 14 being

arranged completely inside a clutch space that is formed by the common disc carrier for both clutches 14, 15. The discs of clutch 15 can be arranged in three-dimensional terms radially above the discs of clutch 14, but also axially next to the discs of clutch 14. Brakes 04 and 05, arranged axially between the two clutches 15 and 24 in the example of Fig. 6, can also, in a different component arrangement configuration, be arranged in three-dimensional terms radially above the planetary gear sets.

[048] In all embodiments of the transmission according to the present invention, freewheels can additionally be inserted at any point in the transmission, e.g. between a shaft and the housing or between two shafts in order to divide a shaft in two. For example, a freewheel additionally inserted between shaft 3 and housing G serves to assist brake 03, a freewheel additionally inserted between shaft 4 and housing G to assist brake 04, and a freewheel additionally inserted between shaft 5 and housing G to assist brake 05. This insertion of an additional freewheel parallel to a shifting element can be provided, for example, in order to improve shifting smoothness when downshifting (coasting shifts).

[049] In addition, the input drive shaft can be separated from the engine by way of a clutch element; the clutch element can be embodied, for example, as a dry or wet initial movement clutch, a magnetic powder clutch, a centrifugal clutch, a hydrodynamic clutch, etc.

[050] The input drive shaft can furthermore also be separated from the engine by way of a converter element; the latter can be embodied as a hydrodynamic converter, a differential converter, an initial movement retarder, a hydrostatic transmission, an electrical linkage or electromechanical linkage, or the like. This means that an additional conversion ratio stage, having a constant conversion ratio or also a variable ratio greater than or equal to unity, can be provided between the engine and transmission.

[051] Alternatively, an initial movement element can also be arranged after the transmission in the power flow direction, so that the input drive shaft is connected in fixed fashion to the coupling shaft of an engine. In such a case, initial movement

is implemented by way of a shifting element of the transmission, e.g. by means of brake 04, which is activated both in the first forward gear and in reverse gear.

[052] In addition, a wear-free brake, for example a hydraulic or electric retarder or the like, can be arranged on each shaft, but preferably on input drive shaft 1 or output drive shaft 2.

[053] It is moreover possible, because of the design according to the present invention, to arrange input drive AN and output drive AB both for transverse, front longitudinal, rear longitudinal, or all-wheel arrangements on the same side of the transmission or of transmission housing G, and for standard input drives on the opposite sides of the transmission or of transmission housing G. An axle differential and/or a center differential can additionally be arranged on the input drive side or the output drive side. Input drive AN and output drive AB of the multiple-ratio transmission according to the present invention can thus be arranged coaxially or non-coaxially (e.g. axially parallel or at an angle) with respect to one another.

[054] A power takeoff for driving additional accessories can furthermore be provided on any shaft, but preferably on input drive shaft 1 or output drive shaft 2.

[055] The shifting elements themselves advantageously comprise on-load shifting clutches or brakes, such as multi-disc clutches, band brakes, cone clutches, or the like; they can also, however, comprise positive-engagement clutches or brakes, for example dog clutches or synchronizers.

[056] A further advantage of the multiple-ratio transmission presented here is the fact that an electrical machine can additionally be mounted on any shaft as a generator and/or as an additional input drive machine.

[057] Any design embodiment, in particular any three-dimensional arrangement of the planetary gear sets and the shifting elements, of itself and in combination and to the extent technically appropriate, of course also falls within the scope of protection of the present claims without influencing the function of the transmission as recited in the claims, even if those embodiments are not explicitly presented in the Figures or in the description.

Reference numerals

1 Shaft

2 Shaft

3 Shaft

4 Shaft

5 Shaft

6 Shaft

03 Brake

04 Brake

05 Brake

14 Clutch

15 Clutch

24 Clutch

45 Clutch

P1 Planetary gear set

P2 Planetary gear set

P3 Planetary gear set

i Conversion ratio

G Housing

AN Input drive

AB Output drive